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RM No. SE8A22

SE 8A22

RESEARCH MEMORANDUM

for the

Air Materiel Command, Army Air Forces

PERFORMANCE OF COMPRESSOR OF XJ-41-V TURBOJET ENGINE

V - PERFORMANCE ANALYSIS OF COMPRESSOR WITH REVISED VANED COLLECTOR OVER RANGE OF COMPRESSOR SPEEDS FROM 3600 TO 11,500 RPM

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

WASHINGTON

JANUARY 23 1948

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SUMMARY

An investigation of the XJ-41-V turbojet-engine compressor with a revised vaned collector was conducted to determine the performance of the compressor and to obtain fundamental information on the aerodynamic problems associated with large centrifugal compressors of this type. The original vaned collector was revised by increasing the flow area at the vaned collector entrance.

A maximum adiabatic efficiency of 0.81 was obtained at a corrected weight flow of 36.5 pounds per second and a pressure ratio of 1.90. The peak pressure ratio was 3.93 and occurred at an impeller speed of 11,500 rpm at a corrected weight flow of 65.5 pounds per second. Revision of the vaned collector resulted in an increased air-flow capacity over the speed range. The design air-flow capacity of 78 pounds per second was very nearly reached at the engine design speed of 11,500 rpm.

The compressor air-flow choking point occurred in the vaned collector passage; however, at speeds above 8000 rpm, the air-flow capacity of the impeller was being approached as indicated by large pressure losses in the impeller at maximum air-flow conditions. An increase in compressor air-flow capacity at the higher speeds can possibly be obtained by removal of the flow restriction in the impeller, which would result in an increased air density at the vaned collector entrance.

INTRODUCTION

At the request of the Air Materiel Command, Army Air Forces, an investigation of the XJ-41-V turbojet-engine compressor was conducted at the NACA Cleveland laboratory. The objectives of this investigation are to determine the performance of the compressor over a range of speeds and weight flows and to obtain fundamental information on the aerodynamic problems associated with large centrifugal compressors.

Information on the cause of the air-flow restriction in the compressor as well as the analysis of the air flow through the compressor is presented in references 1 and 2. This analysis indicated that the maximum flow limitation of the compressor was due to flow separation at the entrance to the vaned collector with a resulting decrease in effective area available to the flow. As a result of this analysis, the flow area at the vaned collector entrance was increased to obtain larger mass flows. The area increase was obtained by cutting back the entrance edges of the collector vanes, which resulted in an increased vaneless-diffuser radius. A preliminary report on the performance of the revised compressor at equivalent speeds of 5000, 7000, 8000, and 9000 rpm is made in reference 3.

The results of the research conducted on the revised compressor are reported herein. Over-all compressor performance data are presented for equivalent speeds of 3600, 5000, 7000, 8000, 9000, 10,000, and 11,500 rpm. The maximum mass flows obtainable over the speed range for both the original and revised compressors are compared. The performance of the impeller and diffuser as separate components is analyzed and information on the location of the air-flow restriction in the compressor is included. The variations in the flow angle of the air at the vaned collector entrance, in the vaned collector passage, and at the simulated burner entrance are discussed.

APPARATUS AND PROCEDURE

Compressor

The compressor assembly is described in reference 4. The impeller for the revised compressor was slightly altered by increasing the blade taper at the inlet and by rounding the inlet blade tips in order to increase the blade natural frequencies that were found to be critical for the impeller used in the original compressor (reference 5). The vaned collector was altered according to the recommendations presented

in reference 2. The flow area at the vaned collector entrance was increased by cutting back the entrance edges of the collector vanes, which resulted in an increased vaneless-diffuser radius. Figure 1 shows a comparison of the original and revised vaned collector entrances.

Instrumentation

The compressor installation is fully described in reference 4. Standard pressure and temperature stations in the inlet pipe (references 6 and 7) together with 10 total-pressure, 13 static-pressure, and 15 temperature stations located in the simulated burner annulus were used to determine the over-all performance of the compressor. Forty-three static-pressure taps were used to measure the static-pressure variation along the flow path through the compressor, to calculate the impeller adiabatic efficiency, and to calculate the diffuser static-pressure-rise efficiency. Several duplicate static pressure taps noted in reference 1 were eliminated in the installation of the revised compressor because symmetry of flow showed that they were unnecessary. A motor-operated remote indicating survey probe of the Fechheimer type (reference 8) was mounted on the compressor front cover and was used to measure the angularity and total pressures of the air stream entering a typical vaned collector passage. This probe was installed normal to the air-flow passage walls and approximately 1 inch ahead of the vaned collector passage, the distance being measured along the mean-flow-path line. Angles were measured relative to a radial line extending from the impeller axis. Survey probes of a similar type were installed in a vaned collector passage and at the entrance to the simulated burner annulus for measuring air-stream angles and total pressures.

Procedure

The compressor was operated under the following conditions:

Equivalent speed (rpm)	Inlet		Outlet
	Total pressure (in. Hg abs.)	Temperature ($^{\circ}$F)	Total pressure (in. Hg abs.)
3,600	-------------	Ambient	31
5,000	-------------	Ambient	33
7,000	14	Ambient	-------------
8,000	14	Ambient	-------------
8,000	8	-32	-------------
8,000	8	Ambient	-------------
9,000	14	Ambient	-------------
10,000	14	Ambient	-------------
11,200	14	Ambient	-------------
11,500	6	-32	-------------

Low inlet total pressures were not used at the speeds of 3600 and 5000 rpm because of possible adverse pressure differentials on the impeller casing at these speeds. At the manufacturer's request, an actual impeller speed of 11,500 rpm was set as the maximum allowable speed for this investigation. Inasmuch as no means were available at the time for regulating the inlet-air temperature between the ambient value of approximately 80° F and the laboratory refrigerated-air value of approximately -32° F, a run was made at maximum allowable speed with 80° F inlet-air and at an equivalent speed of 11,500 rpm with the inlet-air temperature of -32° F. Only one point (maximum flow) was run under ambient-temperature conditions and the resulting equivalent speed was 11,200 rpm. Because of mass-flow limitations of the refrigerated-air supply, it was necessary to reduce the inlet pressure to 6 inches of mercury absolute when using a temperature of -32° F. The equivalent speed of 8000 rpm was investigated at three inlet conditions to correlate the effects of inlet conditions on the compressor performance. Surveys of the air stream were made at the vaned collector entrance and in the vaned collector passage at all speeds except 7000 and 9000 rpm, and similar surveys were made at the entrance to the simulated burner annulus at all speeds.

Rating Methods

The performance of the compressor was based on the measured total pressures and temperatures at the impeller entrance and at the exit of the simulated burner annulus. Separate determinations were made of impeller and diffuser performance. The impeller and diffuser were rated according to the methods described in reference 2.

RESULTS AND DISCUSSION

Compressor Performance

The compressor performance characteristics over the range of equivalent compressor speeds from 3600 to 11,500 rpm are shown in figure 2. A maximum adiabatic efficiency η_{ad} of 0.81 was obtained at a corrected weight flow of 36.5 pounds per second and a pressure ratio of 1.90. The peak pressure ratio was 3.93 and occurred at an impeller speed of 11,500 rpm at a corrected weight flow of 65.5 pounds per second and with an adiabatic efficiency of 0.72. The maximum corrected weight flow at 11,500 rpm was 76 pounds per second.

The compressor maximum corrected weight-flow variations with equivalent impeller speed for the original (reference 2) and revised compressors are shown in figure 3. Data are not presented for speeds

lower than 7000 rpm inasmuch as choking conditions were never reached in the compressor for these speeds because of capacity limitations of the exhauster system under these operating conditions. The curves of figure 3 show that a considerable increase in maximum corrected weight flow was obtained with the revised compressor over the speed range available for the compressor. The largest increase, 27 percent, occurred at 7000 rpm and the increase at the highest comparative speed, 10,000 rpm, was 12 percent. Extrapolation of the revised compressor-flow curve indicated that at an equivalent speed of 11,500 rpm the design flow of 78 pounds per second would almost be reached. The data point at 11,500 rpm, however, for an inlet-air temperature of -32° F and an inlet total pressure of 6 inches of mercury absolute was approximately 2.5 percent lower than the extrapolated value for this speed.

In order to determine some of the effects of inlet conditions on compressor performance, an investigation was made at an equivalent impeller speed of 8000 rpm for different inlet temperatures and pressures. Figure 4 shows the variation of compressor adiabatic efficiency and pressure ratio with weight flow for three different inlet conditions. No appreciable change in pressure ratio was noted with change in inlet conditions but a definite drop in efficiency was observed when the lower inlet temperature was used. This reduction in efficiency has been noted by other investigators (reference 9) and is generally attributed to the effect of increased heat transfer. Changing the pressure of the inlet air from 14 to 8 inches of mercury absolute at constant temperature had a negligible effect on compressor efficiency. The maximum weight flow was decreased by approximately 3 percent when both the inlet-air pressure and temperature was reduced and by approximately 2 percent when the inlet pressure alone was reduced. These values are of about the same magnitude as that mentioned previously for 11,500 rpm. In general, a change in inlet pressure had more effect on maximum weight flow than a change in inlet temperature, and a reduction in either resulted in slightly lowered values of the air flow.

Compressor Components

<u>Component performance.</u> - The performance of the revised compressor and its components, the impeller and diffuser, is shown in figure 5. At the lower impeller speeds from 3600 to 8000 rpm, the impeller efficiencies were high and had a comparatively constant value over the flow range, with a maximum value of 0.95 being reached at 3600 rpm. At higher speeds from 8000 to 11,500 rpm, the impeller performance curves fell off considerably with increased weight flow, with an efficiency of 0.62 occurring at maximum flow at the design

speed of 11,500 rpm. The rapid decrease in impeller efficiency at high-flow, high-speed conditions indicated that pressure losses preliminary to choking in the impeller were being encountered.

In general, the peak diffuser efficiency occurred at approximately the center of the flow range at each speed. Peak diffuser efficiencies varied somewhat over the speed range, a maximum value of 0.78 occurring at 11,500 rpm. At similar values of air flow, the revised compressor generally had higher efficiencies than the original compressor and the useful range of operation of the revised compressor was shifted to higher values of air flow, which showed that altering the vaned collector resulted in improved matching of the impeller and diffuser components. The rapid decrease in diffuser efficiency at high air flows indicated that choking was approached in this component.

Flow capacity and limitations. - The static-pressure variation through the compressor is shown in figure 6 for peak-efficiency and maximum weight flow at equivalent impeller speeds of 8000, 10,000, and 11,500 rpm. For peak-efficiency, a steady pressure rise in general occurred through the impeller, vaneless diffuser, and vaned collector. At 11,500 rpm, a sharp increase in static pressure occurred in the vaned collector between stations 17 to 20, (fig. 6(c)) which indicates that at the peak-efficiency flow this part of the vaned collector was performing very efficiently. Very little additional static-pressure rise occurred in the rest of the vaned collector at this operating condition. Inasmuch as the static-pressure stations were uniformly spaced along the flow path through the collector, the most effective conversion of velocity pressure to static pressure evidently occurred in approximately the first 30 percent of the passage length.

The curves show that for maximum flow a pressure loss occurred near the impeller inlet and in the vaned collector. The largest drop occurred in the vaned collector and, as in the original compressor (reference 2), was a result of choking in this component. The choking occurred inside the vaned collector, and the minimum pressure was reached approximately one-third the distance of the flow path through the collector. In general, about 50 to 65 percent of this static-pressure loss in the vaned collector was recovered by the time the air had reached the burner annulus.

A study of the maximum-flow curves in figure 6 shows that with an increase in speed the pressure drop near the impeller entrance became larger and that at speeds above 8000 rpm the air-flow capacity of the impeller was being approached. At 11,500 rpm, compression in the impeller started with a static pressure approximately 40 percent below the static pressure at the impeller inlet.

This undesirable pressure loss was caused by the flow capacity of the impeller being approached and could possibly be eliminated by cutting back the entrance edge of the impeller blades to give an impeller of increased design air-flow capacity. Reference 10 shows that the impeller air-flow capacity is dependent upon the impeller-inlet design. The removal of this flow restriction would result in higher pressures at the impeller discharge with an accompanying increase in compressor efficiency. In addition, the increased pressure rise through the impeller would increase the air density at the vaned collector entrance, which would result in a probable increase in the air-flow capacity of the vaned collector.

Air-stream angular surveys. - The results of the air-flow-angle surveys near the entrance and inside of the vaned collector and at the entrance to the simulated burner annulus are shown in figure 7. The results of the vaned-collector-entrance surveys are plotted in terms of the angle of attack on the vane (fig. 7(a)), where the angle was considered negative when the air flowed from the high- to the low-pressure side of the vane. Inasmuch as the blade angle was 18°, an angle of attack of -18° represents absolute tangential flow. With the exception of the data at an equivalent impeller speed of 11,500 rpm, the curves show the angle of attack increasing from the outer to the inner wall. The maximum-weight-flow curves generally have a more positive angle of attack than the peak-efficiency curves although at 10,000 rpm the difference was small. At an equivalent impeller speed of 11,500 rpm and at peak compressor efficiency, the average angle of attack across the passage was very nearly zero, a slightly positive angle being obtained near the center of the passage but changing to slightly negative values near the inner and outer walls (fig. 7(a)). The air-flow distribution and air-stream angle of flow relative to the collector vane approached the desired inlet flow conditions. The effect of good vane inlet conditions are shown in figure 6 where at a speed of 11,500 rpm the static-pressure rise in the vaned collector near the vane entrance was much greater than for corresponding peak-efficiency conditions for all other speeds presented.

The results of the survey across a vaned collector passage approximately midway between the inlet and the discharge of the collector show that the flow angle was lower on the inner wall than on the outer wall (fig. 7(b)). Because a flow angle of 90° would represent absolute tangential flow, it is evident that near the outer wall there existed a large tangential component of velocity. With the exception of the data at 5000 rpm, there was little change in flow-angle distribution with change in air flow. At 11,500 rpm, the variation in flow angle from wall to wall amounted to approximately 30°.

The flow-angle variation at the simulated burner entrance showed that considerably smaller tangential angles were obtained on the inner wall than on the outer wall (fig. 7(c)). The tangential angle is the angle between the impeller axis and the vector representing the resultant of the axial- and tangential-velocity components. The magnitude of the angles shows that considerably more tangential than axial flow was present and that in passing through the rear half of the vaned collector no appreciable reduction in the tangential angle was obtained. At the higher impeller speeds, the curves for the two flow conditions tended in general to become coincident.

SUMMARY OF RESULTS

From a performance investigation of the XJ-41-V turbojet-engine compressor with the revised vaned collector over an equivalent impeller speed range from 3600 to 11,500 rpm, the following results were obtained:

1. A maximum adiabatic efficiency of 0.81 was obtained at a corrected weight flow of 36.5 pounds per second and a pressure ratio of 1.90. The peak pressure ratio was 3.93 and occurred at an impeller speed of 11,500 rpm at a corrected weight flow of 65.5 pounds per second.

2. Revision of the vane collector resulted in an increased air-flow capacity over the speed range. The design air-flow capacity of 78 pounds per second was very nearly reached at a speed of 11,500 rpm.

3. Decreasing the compressor-inlet temperature decreased the compressor efficiency and the maximum weight flow obtainable. Decreasing the compressor-inlet pressure had no effect on the compressor efficiency but did decrease the maximum weight flow obtainable.

4. The impeller rated as a separate component had relatively flat efficiency curves over the lower speed range. At the higher speeds, the impeller performance curves fell off with increased weight flow, with efficiency of 0.62 occurring at maximum flow at the design speed of 11,500 rpm.

5. The diffuser rated as a separate component had efficiencies lower than the impeller efficiencies except at a speed of 11,500 rpm. The matching of the revised diffuser and impeller was considerably better than that noted for the original compressor.

6. The compressor air-flow choking point occurred in the vaned collector passage; however, at speeds above 8000 rpm the air-flow capacity of the impeller was being approached as indicated by large pressure losses in the impeller at maximum air-flow conditions. An increase in compressor air-flow capacity at the higher speeds can possibly be obtained by removal of the flow restriction in the impeller, which would result in an increased air density at the vaned collector entrance.

7. Angular surveys at the entrance to the vaned collector showed that an almost constant air-stream angle of attack relative to the vane entrance edge existed across the passage at peak compressor efficiency for a speed of 11,500 rpm. Generally, at all other speeds a large variation in the air-stream angle of attack existed across the passage at the vane entrance.

8. Angular surveys of the air stream in the vaned collector passage and at the entrance to the simulated burner annulus showed that, at both locations and for all speeds, considerably more tangential than axial velocity existed, especially at the outer wall.

Flight Propulsion Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, January 22, 1948.

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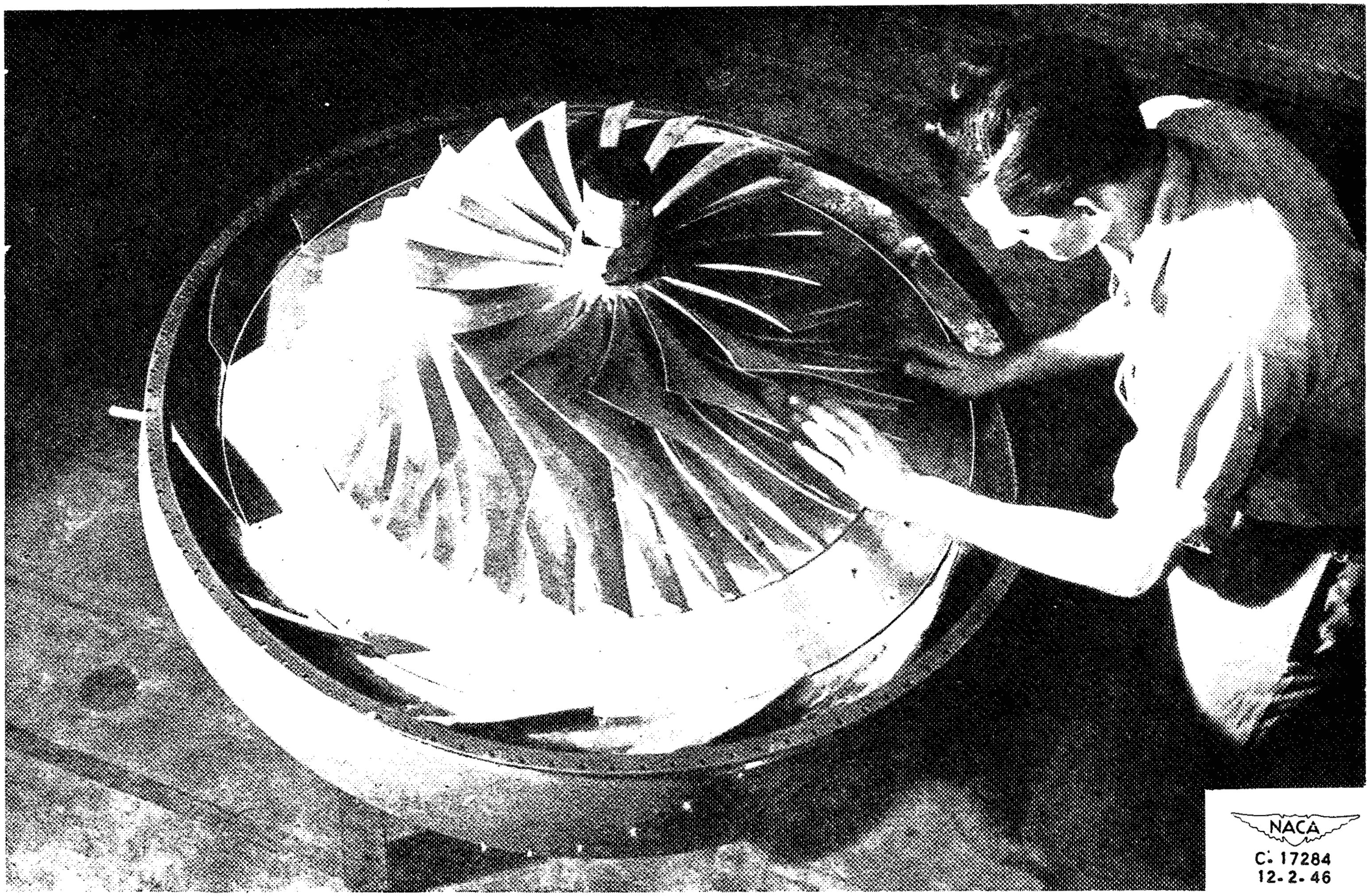

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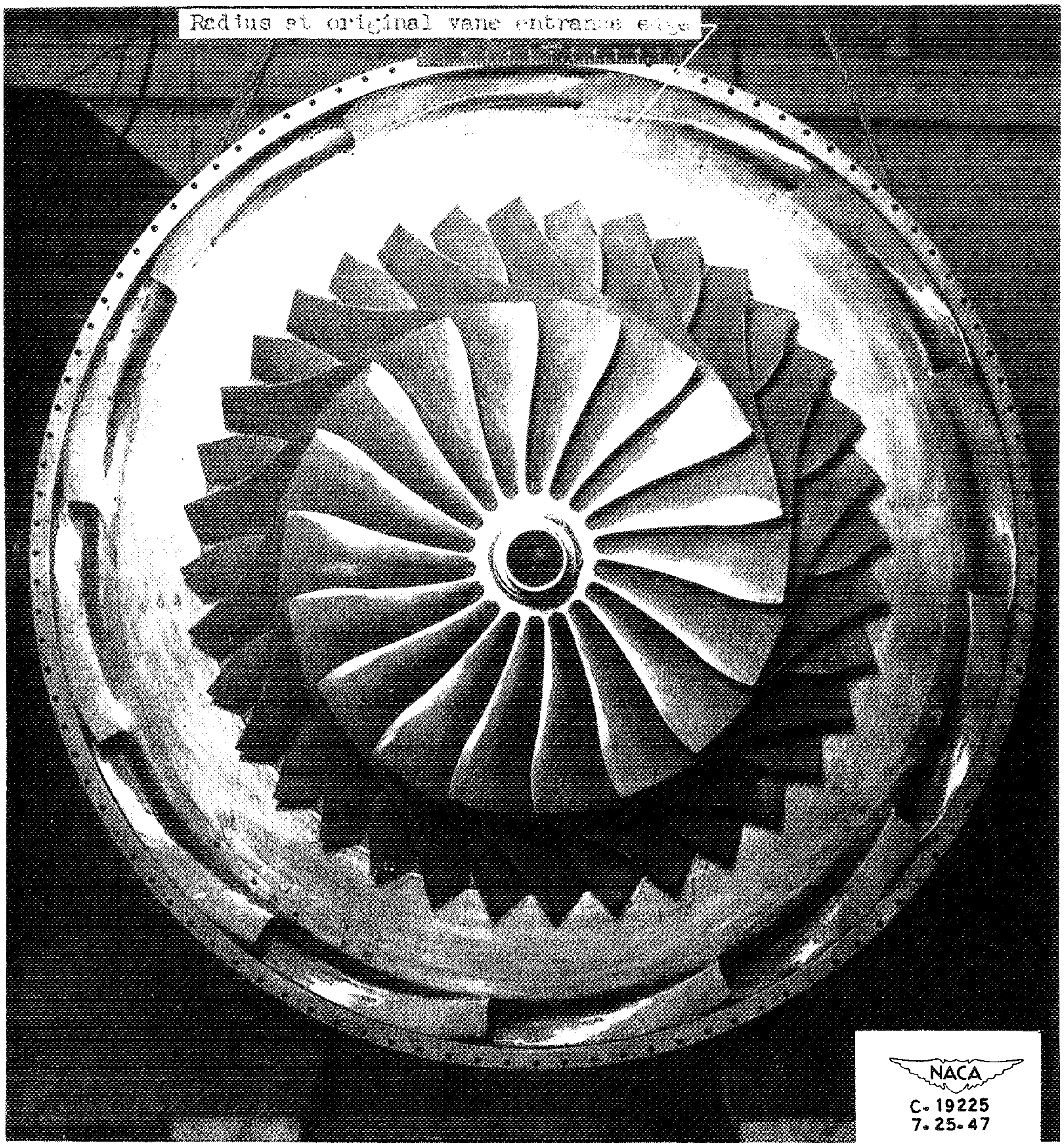
(a) Original compressor.

Figure 1. - View of impeller and vaned collector entrance for original and revised compressors.



(b) Revised compressor.

Figure 1. - Concluded. View of impeller and vaned collector entrance for original and revised compressors.

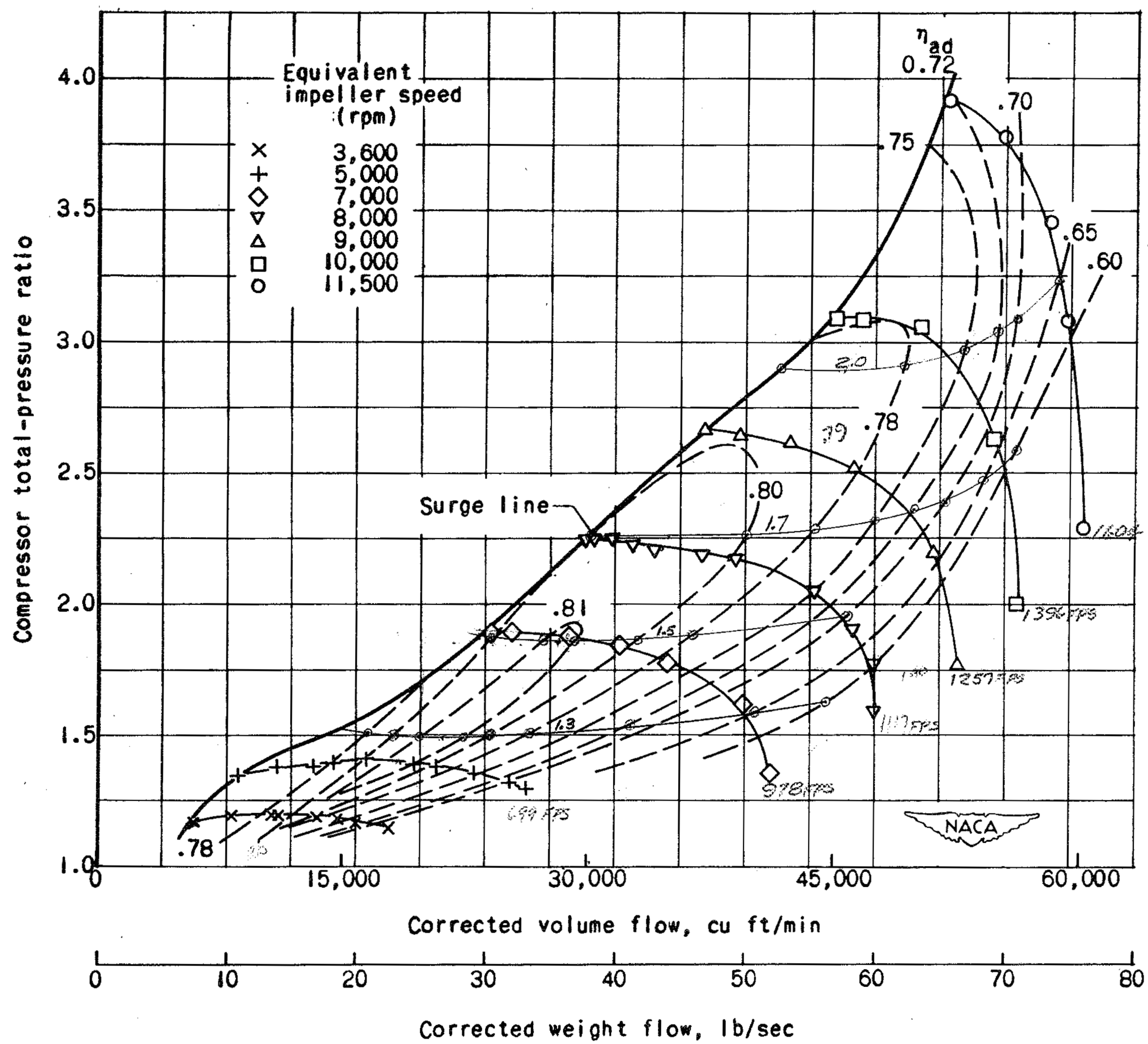


Figure 2. - Compressor performance.

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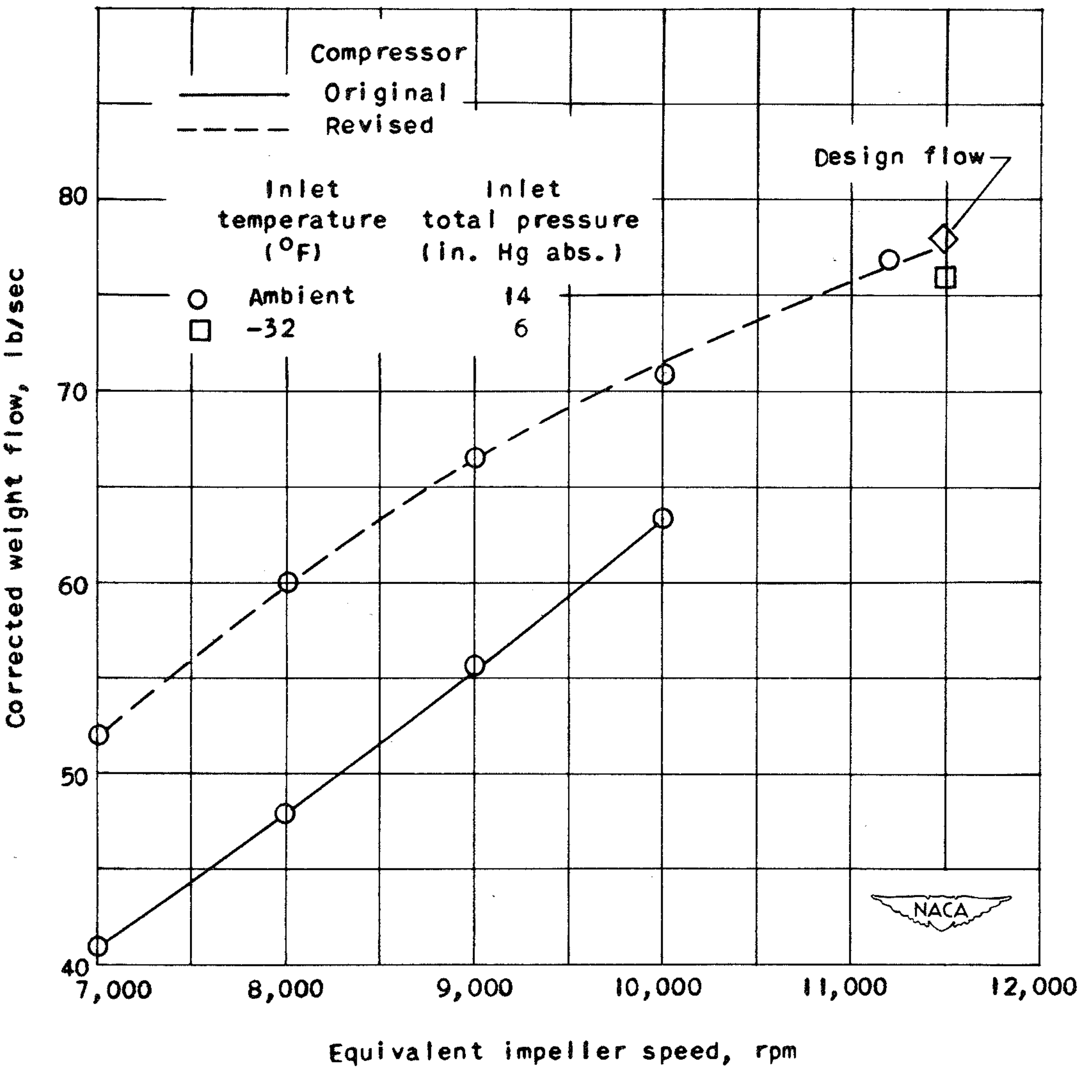

Figure 3. - Effect of compressor revision on maximum weight-flow capacity.

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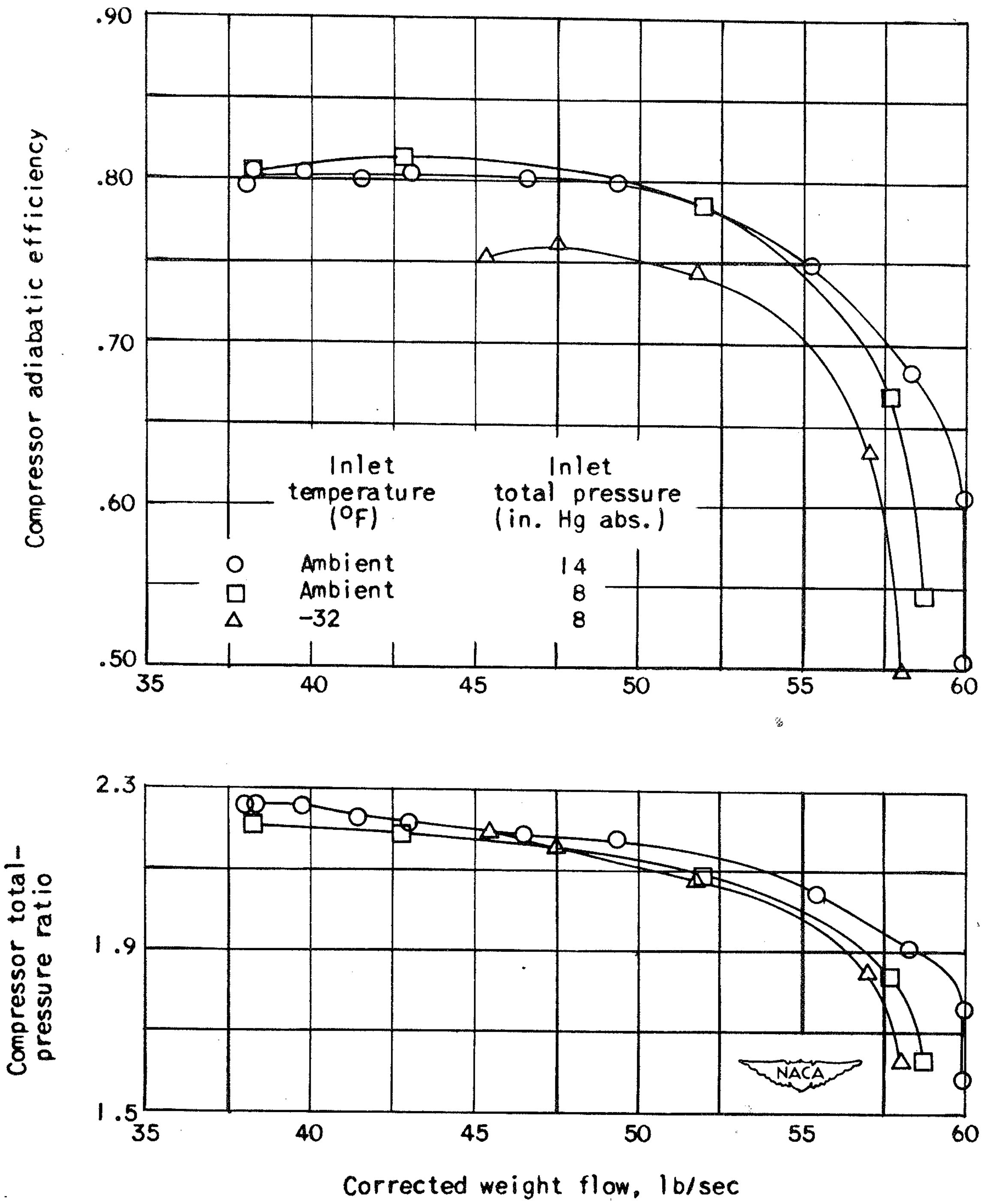

Figure 4. - Effects of varying inlet temperature and pressure on compressor performance for equivalent impeller speed of 8000 rpm.

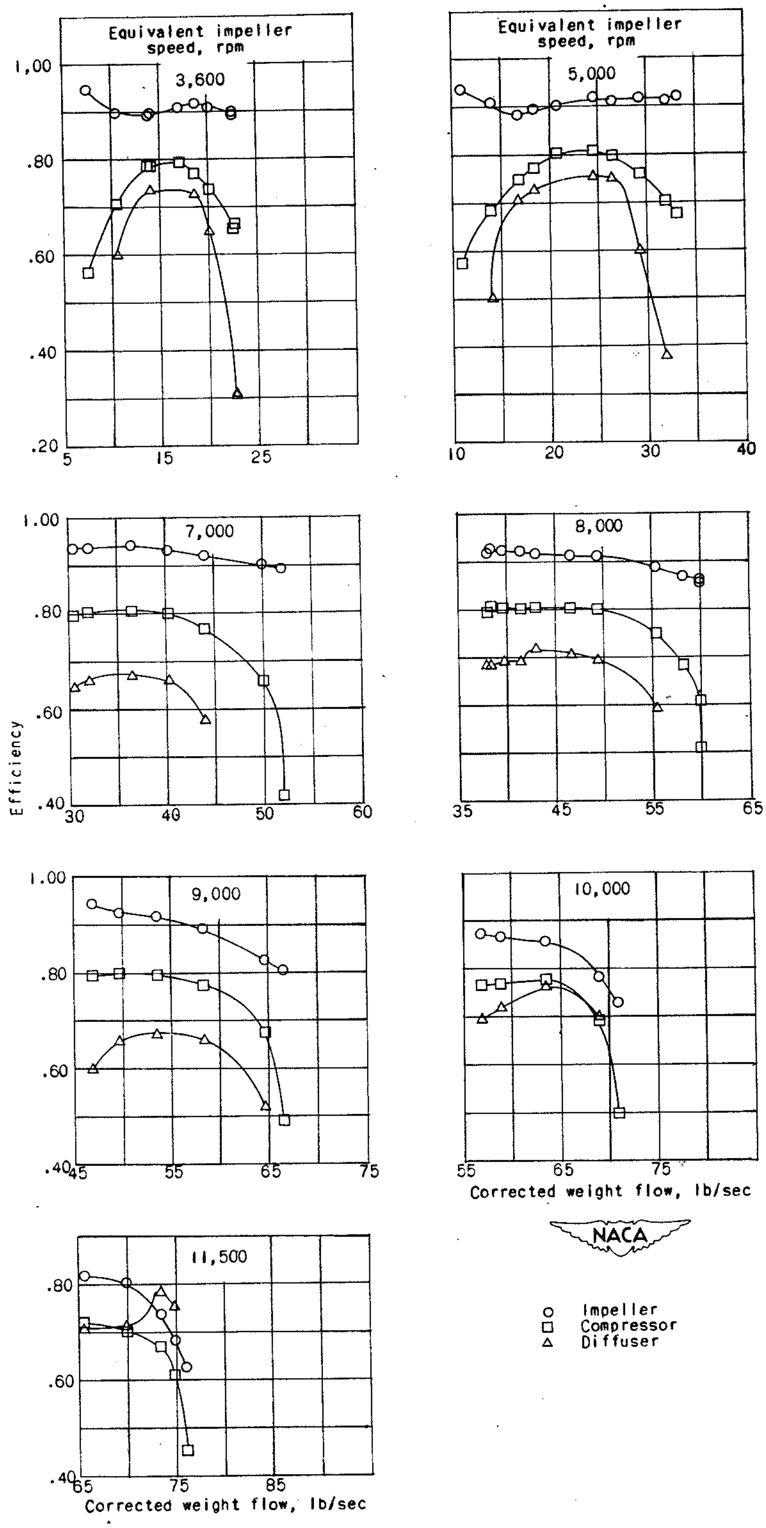

Figure 5. - Performance comparison of compressor, impeller, and diffuser for several impeller speeds.

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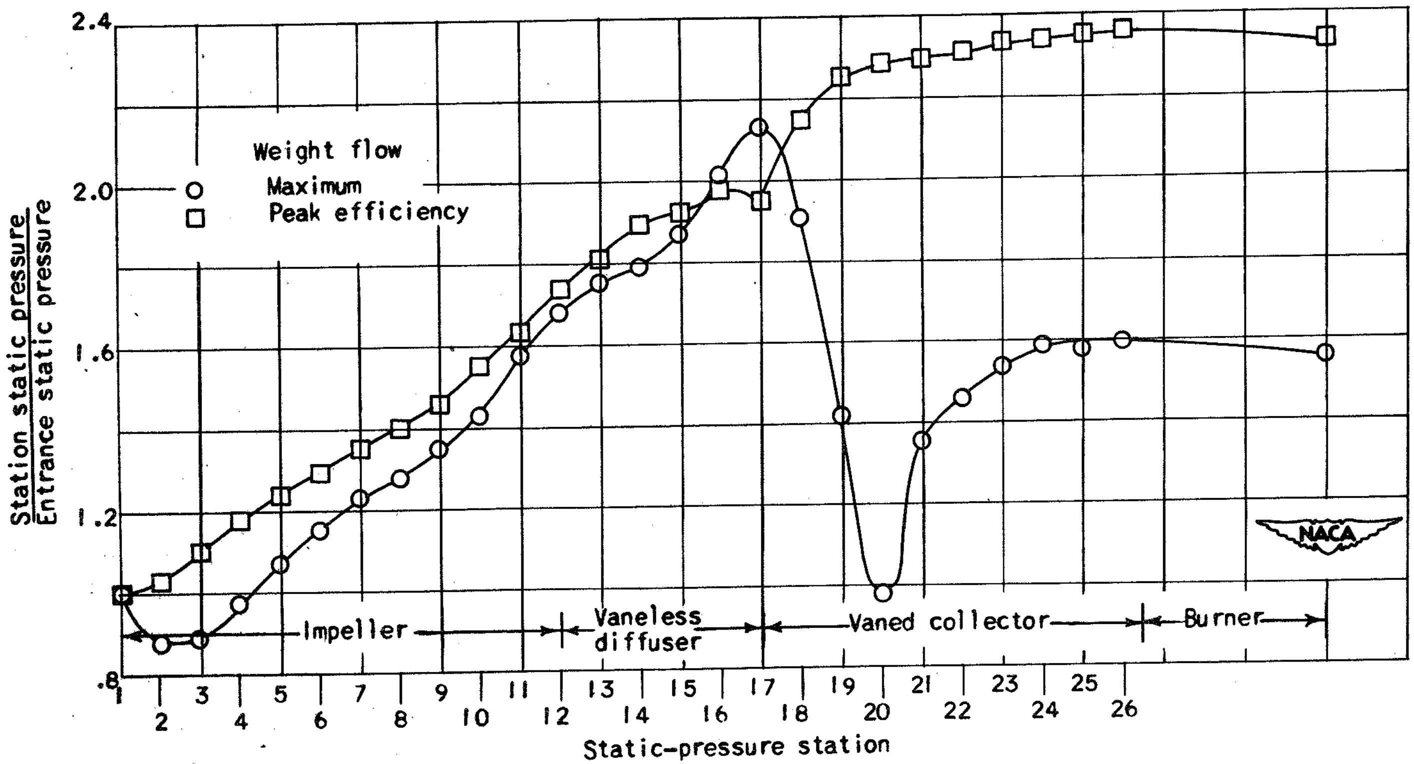
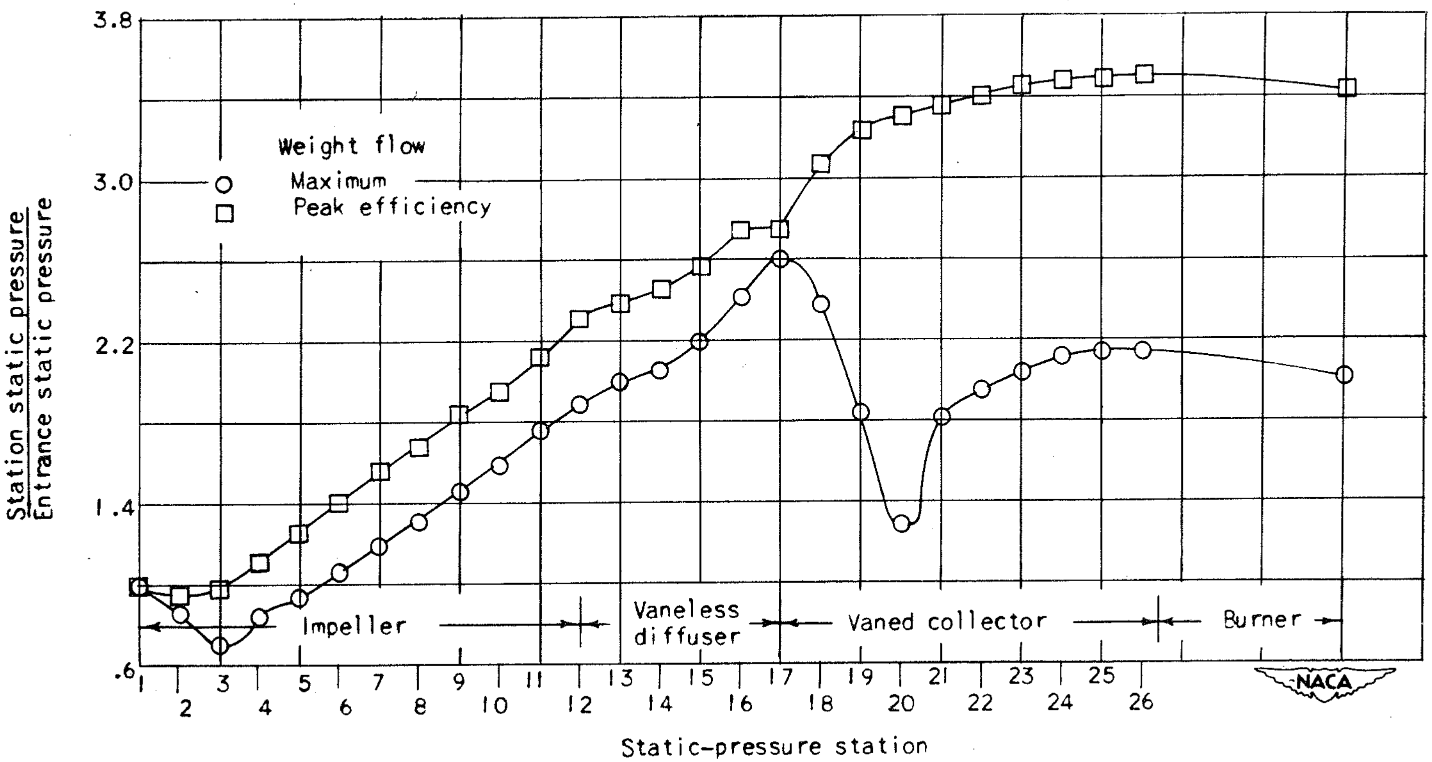


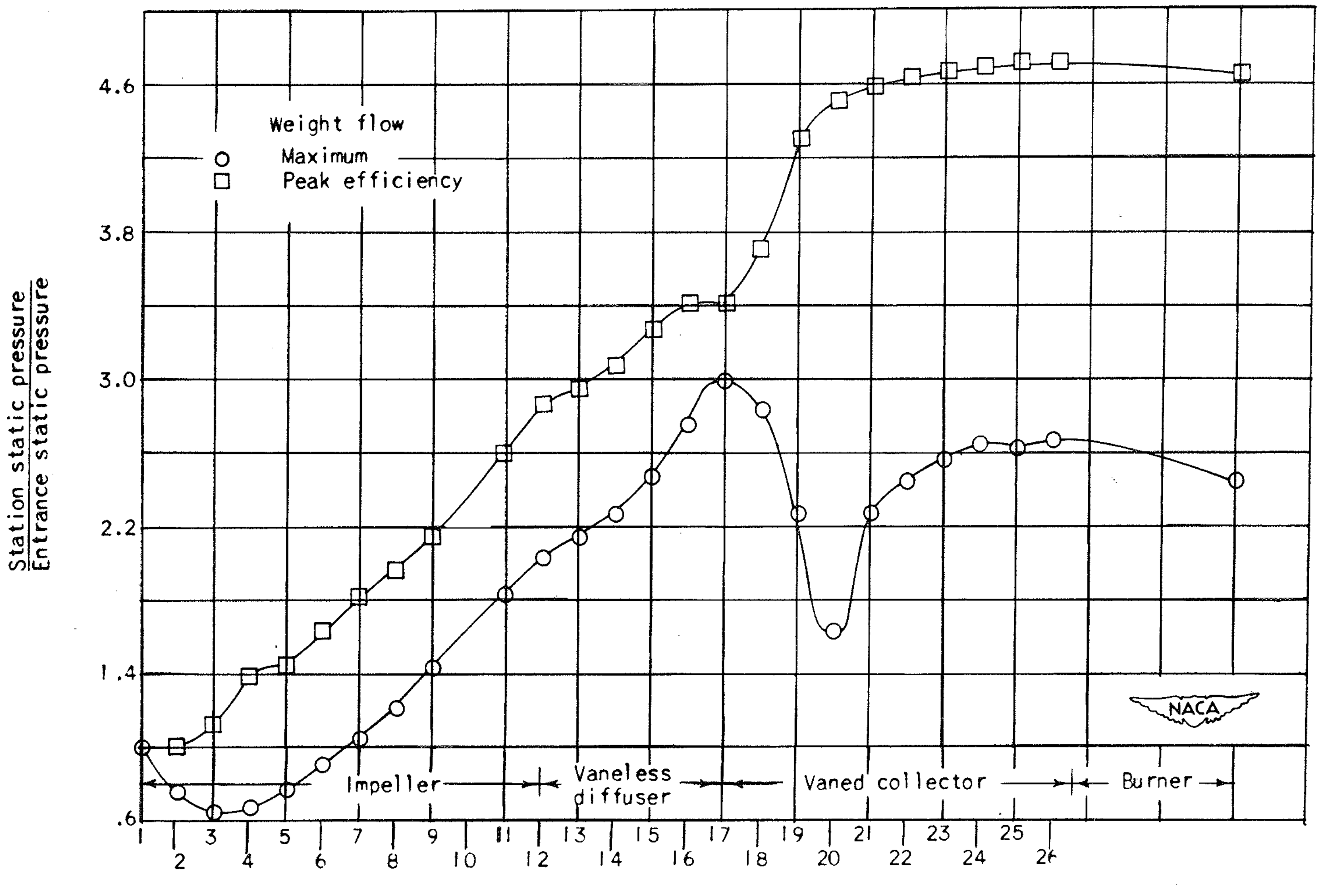
(a) Equivalent impeller speed, 8000 rpm.

Figure 6. - Static-pressure variation through compressor.



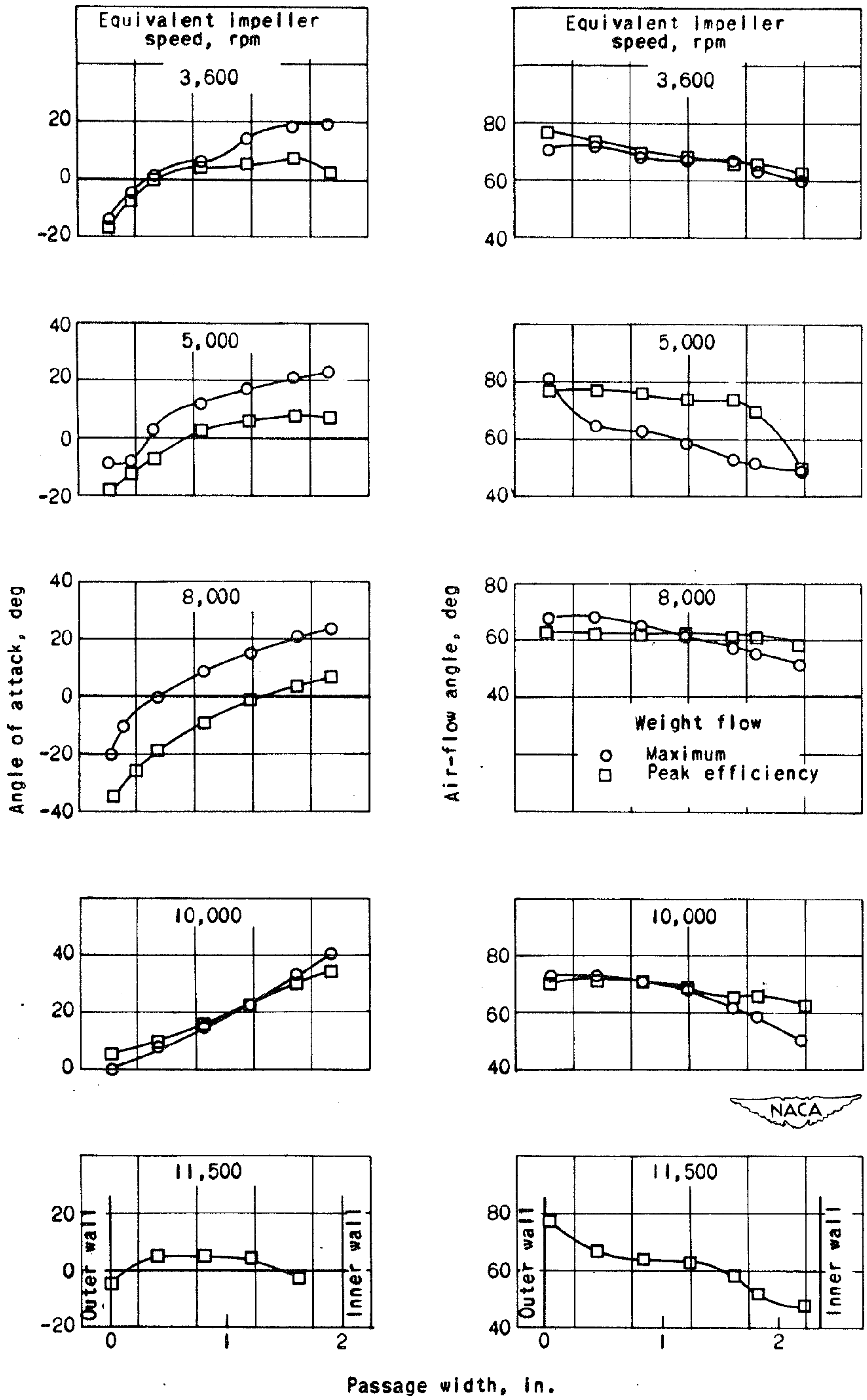
(b) Equivalent impeller speed, 10,000 rpm.

Figure 6. - Continued. Static-pressure variation through compressor.



(c) Equivalent impeller speed, 11,500 rpm.

Figure 6. - Concluded. Static-pressure variation through compressor.



(a) Vaned collector entrance. (b) Vaned collector passage.

Figure 7. - Air-flow-angle surveys.

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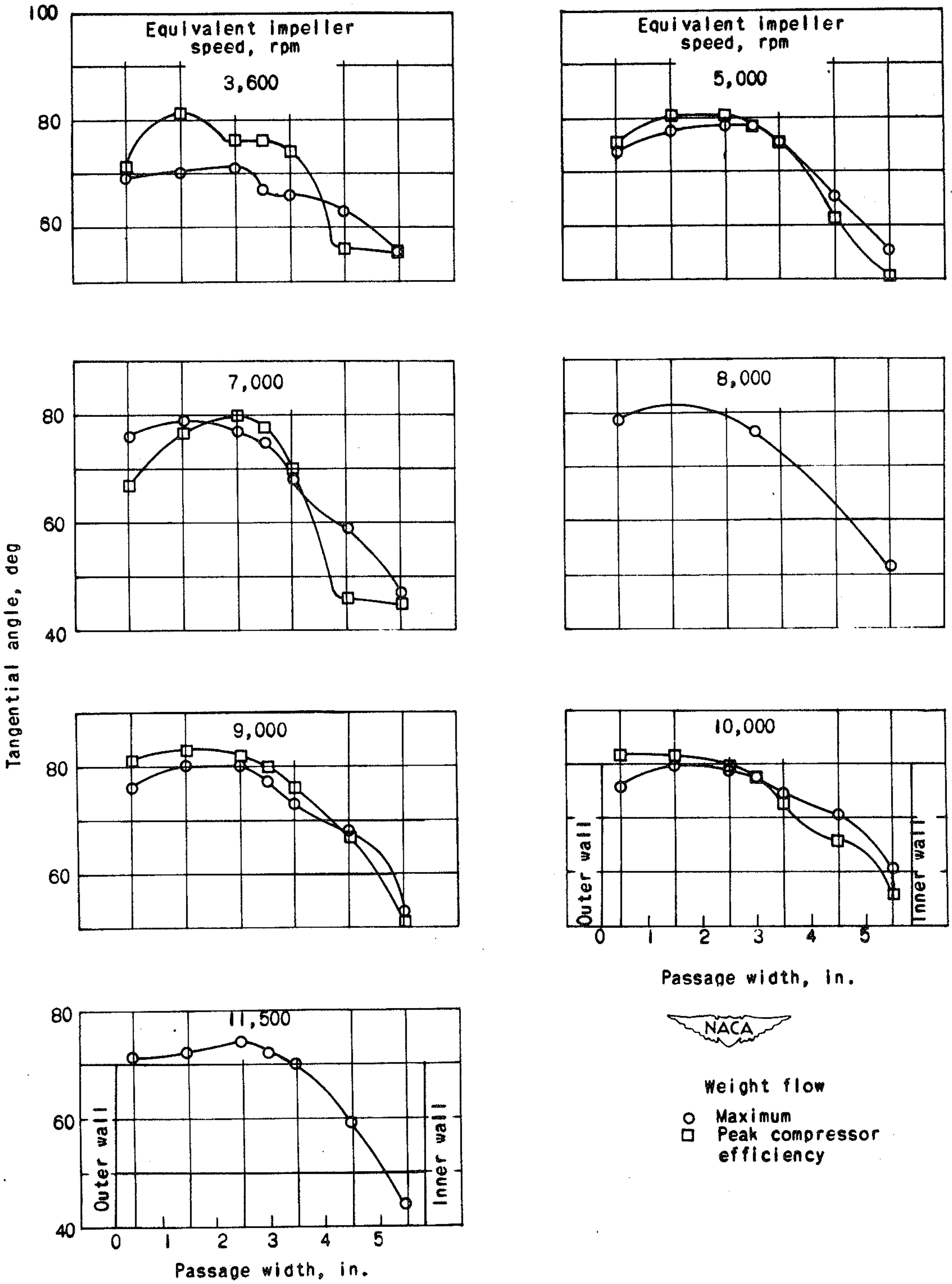

(c) Simulated burner entrance.

Figure 7. - Concluded. Air-flow angle surveys.

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